AD-A041 024 COLD REGIONS RESEARCH AND ENGINEERING LAB HANOVER N H F/G 13/1
DEMONSTRATION OF BUILDING HEATING WITH A HEAT PUMP USING THERMA--ETC(U)
MAY 77 P W SECTOR

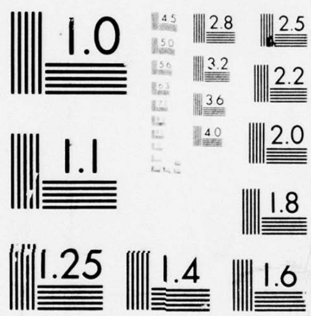
UNCLASSIFIED CRREL-SR-77-11

| OF |

AD
A041024

END

DATE
FILMED
7 - 77



MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

SR 77-11

Special Report 77-11

DEMONSTRATION OF BUILDING HEATING WITH A HEAT PUMP USING THERMAL EFFLUENT

ADA041024

Peter W. Sector

May 1977

DDC
JUN 29 1977
C

AD No.
DDC FILE COPY

CORPS OF ENGINEERS, U.S. ARMY
COLD REGIONS RESEARCH AND ENGINEERING LABORATORY
HANOVER, NEW HAMPSHIRE

Approved for public release; distribution unlimited.

Unclassified

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE

READ INSTRUCTIONS BEFORE COMPLETING FORM

1. REPORT NUMBER: Special Report 77-11

2. GOVT ACCESSION NO.:

3. RECIPIENT'S CATALOG NUMBER: (14) CRREL-SR-77-11

4. TITLE (and Subtitle): (6) DEMONSTRATION OF BUILDING HEATING WITH A HEAT PUMP USING THERMAL EFFLUENT

5. TYPE OF REPORT & PERIOD COVERED:

6. PERFORMING ORG. REPORT NUMBER:

7. AUTHOR(s): (10) Peter W. Sector

8. CONTRACT OR GRANT NUMBER(s):

9. PERFORMING ORGANIZATION NAME AND ADDRESS: U.S. Army Cold Regions Research and Engineering Laboratory, Hanover, New Hampshire 03755

10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS: (16) DA Project 4A762732AT41, (17) Task 6, Work Unit 014

11. CONTROLLING OFFICE NAME AND ADDRESS: Directorate of Military Construction, Office, Chief of Engineers, Washington, D.C. 20314

12. REPORT DATE: (11) May 1977

13. NUMBER OF PAGES: 28 (12) 30p.

14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office):

15. SECURITY CLASS. (of this report): Unclassified

15a. DECLASSIFICATION/DOWNGRADING SCHEDULE:

16. DISTRIBUTION STATEMENT (of this Report): Approved for public release; distribution unlimited.

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report):

18. SUPPLEMENTARY NOTES:

037 100

19. KEY WORDS (Continue on reverse side if necessary and identify by block number):

Heat
Heat pumps
Heating
Refrigeration systems
Waste management

20. ABSTRACT (Continue on reverse side if necessary and identify by block number):

This report describes efforts made to recover waste heat and to reuse it to heat a building. A heat pump, which is a refrigeration device, was operated to provide building heat and to demonstrate both economic benefits and energy savings possible with this type of heating system. Heat pump fundamentals and system design considerations supplement the report of this demonstration project. Operational characteristics were monitored and are reported. A 25% reduction in heating costs was observed compared with an oil-fired system. The author recommends that the minimum coefficient of performance should be 3.4 for a cost- (cont on p. i)

DD FORM 1 JAN 73 1473 EDITION OF 1 NOV 65 IS OBSOLETE

i

Unclassified

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

20. effective, energy-conservative heat pump heating system.

PREFACE

This report was prepared by Peter W. Sector, Mechanical Engineer, of the Applied Research Branch, Experimental Engineering Division, U.S. Army Cold Regions Research and Engineering Laboratory. Funding was provided by DA Project 4A762713AT41, *Design, Construction and Operations Technology for Military Facilities*, Task T6, *Energy Systems*, Work Unit 014, *Low Temperature Heat Sources*.

Technical reviewers of the manuscript were Dr. Haldor W. C. Aamot and Kevin L. Carey.

Thanks are due to both Haldor Aamot and Kevin Carey for their patience and guidance, and especially to Robert Northam and Ronald Farr for their technical support.

The contents of this report are not to be used for advertising or promotional purposes. Citation of trade names does not constitute an official endorsement or approval of the use of such commercial products.

ACCESSION for
NTIS White Section
DDC Buff Section
UNANNOUNCED
JUSTIFICATION
BY
DISTRIBUTION/AVAILABILITY CODES
Dist. AVAIL. and/or SPECIAL
A

Summary of Conclusions and Recommendations

1. Heat pump operation is nearly continuous in space heating applications. Well-designed, reliable components should be emphasized throughout such a system.

2. About 2/3 of the required electrical input energy powers the compressor in a typical heat pump system. The remaining electrical energy powers fans, pumps, and controls. These auxiliary power requirements should be included in any cost-effectiveness appraisal.

3. Heat pumps operate most efficiently and economically within a narrow range of operating conditions. Heat pumps are best suited to situations where source and supply temperatures do not fluctuate significantly.

4. A heat pump heating system is energy conservative if its coefficient of performance exceeds 3.4. This figure should be included in performance criteria.

CONTENTS

LIST OF ILLUSTRATIONS

INTRODUCTION

Industrial process cooling waters are our largest available source of recoverable low temperature heat. A "heat pump" extracts heat energy from some medium such as water, and transfers this heat to a medium at a higher temperature. An efficient heat pump will provide three to four units of heat energy output for each unit of energy input. This report first familiarizes the reader with general concepts and considerations applicable to heat pumps, and then gives the results of actual operations of a typical installation. The objectives of the work were to provide energy-conservative and cost-effective space heating for the building shown in Figure 1, and to operate this system as a thermal waste management demonstration project for the Cold Regions Research and Engineering Laboratory of the Army Corps of Engineers.

Figure 1. Equipment storage and fabrication building, 40 ft (12.2 m) x 100 ft (30.5 m) x 14 ft (4.3 m).

PRECEDING PAGE BLANK-NOT FILMED

A cost-effectiveness evaluation and the resulting recommendations are presented. References are included which contain effective procedures for establishing specifications and design criteria for heat pump systems.

FUNDAMENTALS OF THE HEAT PUMP

The household refrigerator is the most common heat pump. As with this familiar appliance, a heat pump is a mechanical device which removes heat from one medium at a lower temperature and transfers this heat to another medium for delivery at a higher temperature. Heat pumps are now mass-produced for a variety of applications. Reliable, competitively priced, packaged heat pump systems are receiving increased utilization, since rising costs for energy have made heat pump installations attractive alternatives to conventional heating and air conditioning systems. Space heating and air conditioning, water heating, waste heat reclamation and similar functions are common applications of heat pumps. The concepts and considerations discussed here apply to common modes of operation and most standard applications of vapor-compression heat pumps. The emphasis of this report, however, is placed on heat pump utilization to provide building heat.

The Heat Pump Cycle

The thermodynamic cycle used as a basis for a vapor-compression heat pump is the Rankine cycle. Figure 2 is a schematic of the refrigeration components necessary for the operation of this cycle.

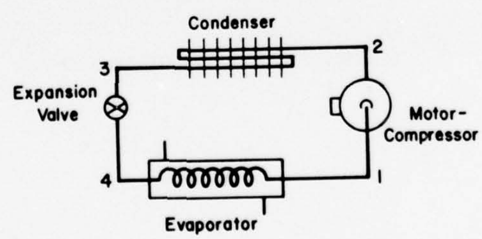


Figure 2. Refrigeration components of a vapor-compression cycle

The cycle follows this sequence:

1 - 2	compressor	isentropic compression
2 - 3	condenser	isothermal condensation at constant pressure and temperature
3 - 4	expansion valve	irreversible expansion at constant enthalpy
4 - 1	evaporator	isothermal evaporation at constant pressure and temperature

Figure 3 presents the pressure-enthalpy chart for this cycle with Refrigerant-22. The use of this chart to determine the performance of an actual operating cycle is demonstrated later in this report (Sample Calculations, App. A).

Major Factors Affecting Heat Pump Performance

The index of cycle operating efficiency is called the coefficient of performance (CP). This is defined, for heating, as the heat output divided by the work input. Usually electricity runs the refrigerant compressor(s), driving the cycle, and also may power accessory equipment such as air blowers, pumps, and components of the control systems. Thus from the total electrical input, operating costs may be determined.

The heat source, from which the refrigerant extracts heat at the evaporator (step 4-1, Fig. 3), may be air, water, earth, or some other medium. This heat is delivered by the condenser (step 2-3, Fig. 3) to the heat sink, in this case a building. For heat to be transferred from the source to the refrigerant and from the refrigerant to the sink, temperature gradients are necessary. Vapor-compression cycles used in heat pumps operate with a 10° to 20°F (6° to 11°C) difference between the refrigerant evaporating temperature and the temperature of the heat source, and with a similar difference between the temperature of the heating medium for the room and the condensing temperature of the refrigerant. These temperature differences depend on the effectiveness of the heat exchangers and bracket the range in which an actual refrigeration cycle must operate.

Pumping heat is analogous to pumping water. The physical system which is described by the increased elevation to which water must be raised may be compared to the increased temperature to which heat must be raised. Pumping either water or heat to higher and higher levels requires more and more work. This input, usually in the form of electrical work which

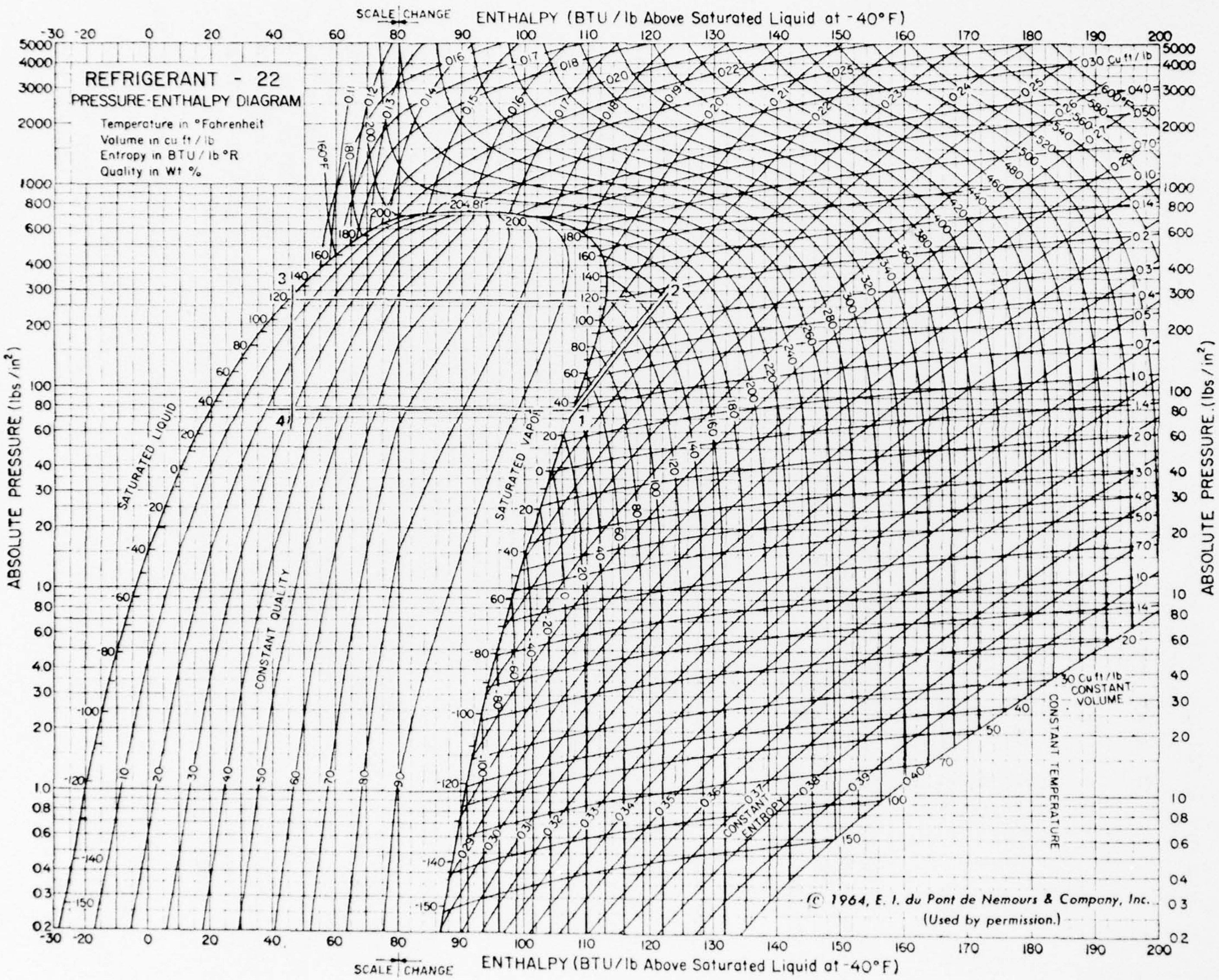


Figure 3. Pressure-enthalpy diagram for Refrigerant 22

drives the heat pump cycle, is what the consumer pays for. Remembering that the coefficient of performance (CP) indicates the units of output of the system for each unit of input, then it is clear that the larger the CP of a heat pump, the less "costly" is its output. A 70-80% efficient unit may be expected to yield a CP of 4.0 or higher, with the practical maximum being 4.5. Losses occur in the compressor, heat exchangers, and all the electrical devices in the system. (A discussion of this inherent inefficiency may be found in references 3, 4, 7, and 10.)

Two important considerations affect heat pump performance. These are stated by Ambrose[3] as:

1. "In an actual heat pump system the CP varies directly as the compressor suction pressure and inversely as the condensing pressure..."

2. "The suction pressure, in turn, is determined by the temperature of the heat source, so that the lower the heat source temperature the lower the compressor suction pressure. Similarly, the head pressure or condensing pressure is determined by the heat sink temperature, i.e. the temperature of the heated medium being circulated to the conditioned space."

Stated more simply, the closer the source temperature is to the delivery temperature, the higher the CP. Using the water pumping analogy, overcoming low head takes less work than overcoming high head for the same flow rate. These two temperature-pressure relationships are interrelated. They establish the operating and performance range of a given unit, and indicate its usefulness in a given system. The following sections report on the analysis, subsequent installation, and operation of a heat pump system as a demonstration project, providing space heating in a maintenance and fabrication building.

DESCRIPTION OF THE DEMONSTRATION PROJECT

In support of the Corps of Engineers Energy Conservation Program, CRREL research engineers developed this heat pump demonstration project. An objective of this project has been to provide all heating needs of the building shown in Figure 1 in an economical and energy conservative manner. Since thermal discharge water is available abundantly from the large refrigeration complex at CRREL, this work was performed to demonstrate the operation of a heating system utilizing "recaptured" or "recycled" thermal effluent. Thus the work supports the Corps of Engineers studies of Thermal Waste Management in Cold Regions.

The building to be heated is a typical industrial type, steel-frame structure, sheathed and roofed with 26-gage steel. It functions as an equipment storage and fabrication building, and as a fabrication area for large projects. This versatile building is representative of many found on military installations, on industrial and construction sites (including the Trans-Alaska Pipeline), and nationwide in applications from barns to warehouses. The CRREL building is 40 ft (12.2 m) wide, 100 ft (30.5 m) long, and 14 ft (4.3 m) high at the eaves. The roof is pitched very slightly. Centered on each short wall is a large vehicle access door, 12 ft (3.7 m) high by 12 ft (3.7 m) wide. Together with a personnel door, these are the only significant openings in the building. Four small fixed windows are located on one long wall, two near each end. There are no interior partitions, the floor is a 6-in. (15 cm) concrete slab, and the inside surfaces of the walls and roof are sprayed with a 2-in. (5 cm) layer of "TCI-75" spray insulation, a cellulose fibrous material. The metal skin and the layer of this insulation have an overall coefficient of heat transmission (U) of 0.085 Btu/h °F ft^2 (0.483 watt/m^2 K).

Use of this facility does not require comfort-heating. Heating systems in central New Hampshire must meet the design temperature of -20°F (-29°C). A temperature difference of 70°F (39°C) was therefore used in heat load calculations, to ensure that the building temperature was held around 50°F (10°C) as a minimum.

Heat Requirements

To estimate the heat requirements, the area method was used. Calculation of the heat transfer of each area, (walls, floor, roof, doors, and windows), showed an estimated heat load of 108,590 Btu/h (31.8 kW) in the worst case of open vehicle access doors at -20°F (-29°C), and a sustained heat load of 64,520 Btu/h (18.9 kW). Another worst case estimate was developed using the infiltration method which assumes that all the air in the building will be replaced and heated in a given time. Consideration of traffic in and out of the building, as well as inspection of the weather seals and the general tightness of the building, indicated two total air changes per hour. In the worst case, as mentioned, this would require 112,560 Btu/h (33.0 kW) of heating. Normal load by this method, calculated with the large doors closed, was estimated at 70,580 Btu/h (20.7 kW), which correlates with the previous estimate. The heat pump capacity specified for this demonstration project was 70,000 Btu/h (20.5 kW).

Utilities System Description

Within its main laboratory, CRREL operates a refrigeration complex with a capacity of 100 tons of refrigeration rejecting 1.2 million Btu/h (352 kW) and discharging condenser cooling water at the rate of

450-500 gallons per minute (0.03 m^3/s) at a nearly constant temperature of 55°F (12.8°C). This clean industrial discharge is taken from a recycling sump and piped via a buried 2-in. (51-mm) diameter PVC line to the heat pump in the building. A bleeder on this supply line runs a small but constant stream into the drain line to provide freeze protection for the supply line under a snow-free parking lot. This water system is sized to meet expansion needs. The heat pump was chosen to operate on a 208-V, 3-phase, 60 Hz electrical power source.

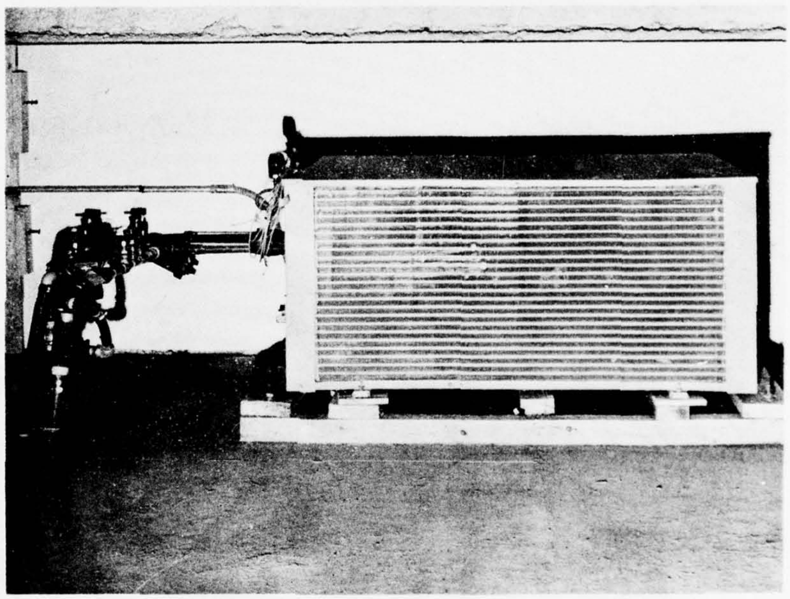


Figure 4. Test stand for operation of the heat pump

Heating System Description

A conventional industrial package chiller, similar to those which provide air conditioning for small office buildings, was chosen because of its availability, simplicity of installation, and in the heat pump mode, predicted economical operation. Using Refrigerant-22, the manufacturer rates this unit at 52,800 Btu/h (15.5 kW) of heat removal (cooling) for 5.7 kW of electrical power consumption. Figure 4 is a photo of the installation. Figures 5 and 6 depict the refrigeration components of the unit. This system extracts heat from the 55°F (13°C) discharge water piped to the evaporator and transfers it to air which is

blown into the interior of the building at about 85°F (30°C). Based on the manufacturer's performance ratings the following calculation provides an estimate of the heating capacity, 72,250 Btu/h (21.2 kW), for this machine:

Given: rated cooling capacity - 52,800 Btu/h (15.5 kW)
power requirement - 5.7 kW

Find: heat of rejection of condenser

5.7 kW (3412 Btu/kWh = 19,449 Btu/h

$$\begin{array}{r} 52{,}800 \\ \underline{19{,}449} \\ 72{,}249 \end{array} \text{ Btu/h}$$

Rated heating capacity - 72,250 Btu/h (21.2 kW).

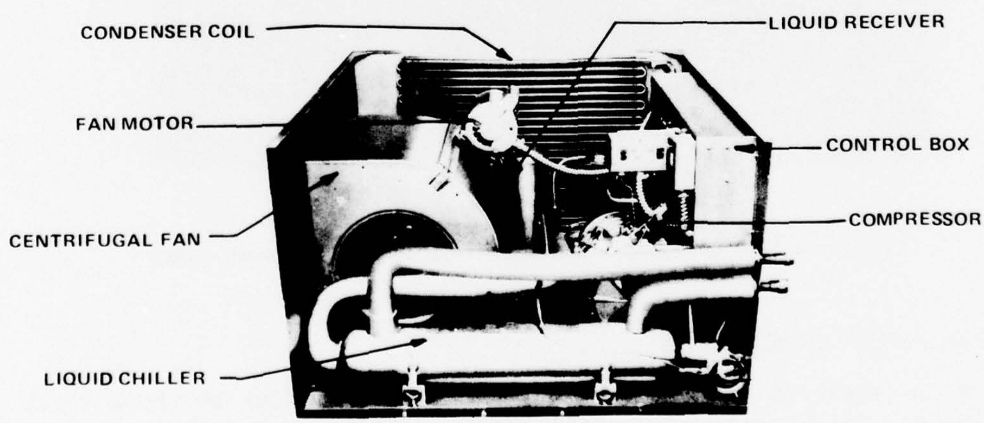


Figure 5. Refrigeration components of the heat pump.

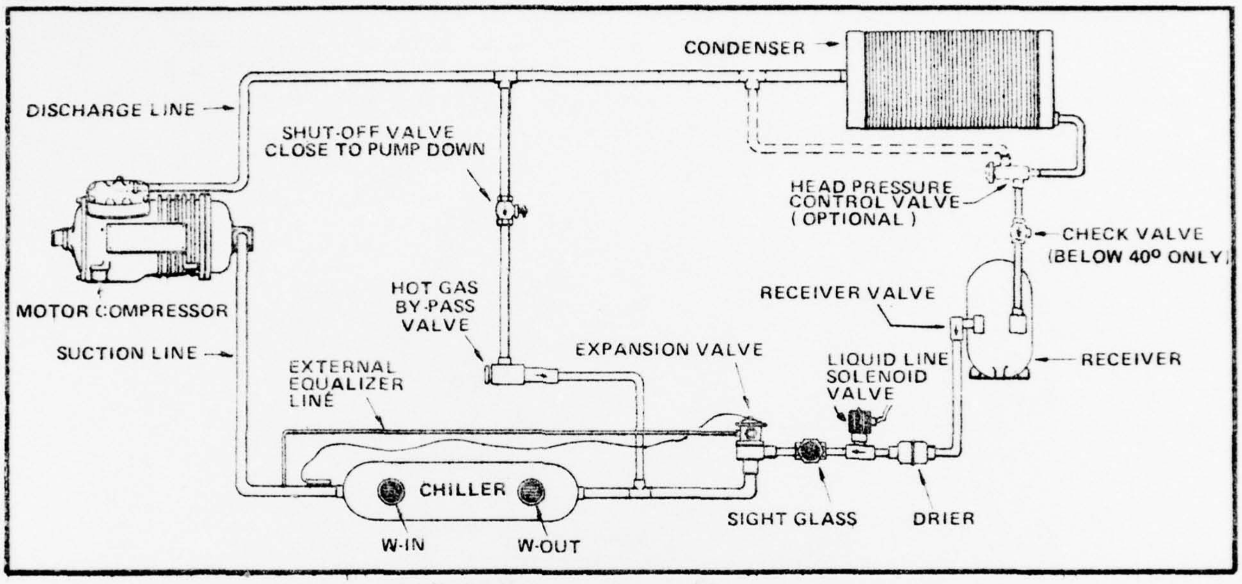


Figure 6. Refrigeration system layout of the heat pump

Instrumentation System and Parameters Observed

The heat pump specified was sized to satisfy the normal heating requirements for the building, ensuring a full range of test conditions as the worst case situation was approached. Mechanical and fluid friction, unwanted heat transfer, motor inefficiencies and other losses contribute to an increase in entropy for all real refrigeration cycles, as indicated by the second law of thermodynamics. Dividing the system into its components can indicate these losses.

The locations of the instrumentation are shown in Figure 7. The flow rate of the water used as the heat source was measured with a Badger Recordall Model 15 water meter. Thermocouples immersed in the water supply and discharge lines indicated the respective water temperatures. Refrigerant temperatures were also recorded at specific points in the cycle to permit adjustment to optimum performance and to gain detailed thermodynamic information about the cycle (ref. 10). All thermocouple readings were taken with a Fluke Model 2100A Digital Thermocouple Thermometer, the couples being made with Thermoelectric copper-constantan thermocouple wire.

Electrical input was recorded with a kilowatt-hour meter, Form 165, made by Sangamo Electric Co. Compressor discharge and suction pressures were monitored with dial gages, as were water supply and discharge pressures. Refrigerant pressures were measured with Robinaire high and low pressure gages; water pressures were read on Ashcroft Model 1000 gages. Time was recorded with a Durham stopwatch. Table I lists all parameters observed and the error associated with the accuracy of each device and the accuracy of the readings.

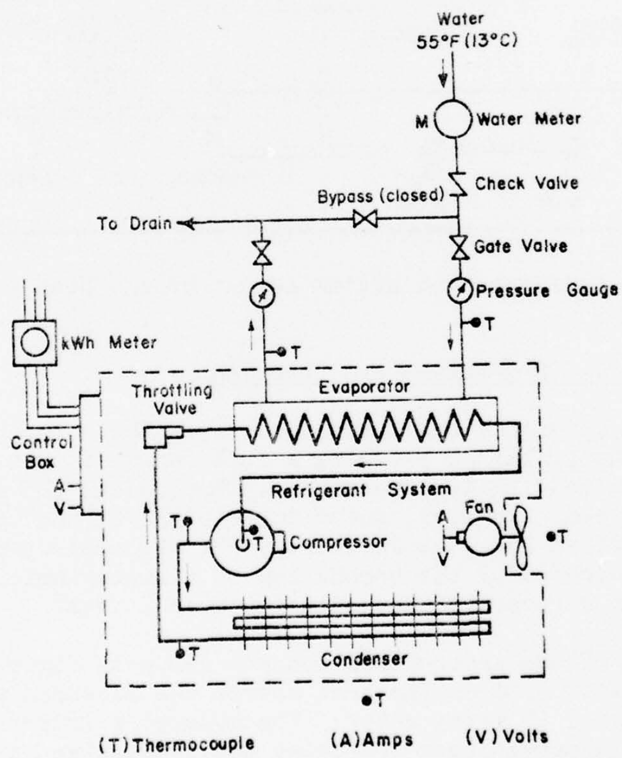


Figure 7. Schematic of instrument locations on heat pump

Table I. Parameters observed and uncertainty (% full scale) associated with the reading.

WATER		
	Supply, discharge	Pressure (psi $\pm$ 3%) Temperature (°F $\pm$ 1.4°) Flow Rate (gal $\pm$ 0.5%)
REFRIGERANT		
	Supply, discharge	Pressure (psi $\pm$ 5%) Temperature (°F $\pm$ 1.4°)
TIME		Minutes ($\pm$ 0.2 s)
ELECTRICAL POWER INPUT		Kilowatt-hours ($\pm$ 0.1 kWh)
FAN POWER INPUT		Volts ($\pm$ 1%) Amps ($\pm$ 2.5%)
AIR TEMPERATURES		Outside ambient (°F $\pm$ 1.4°) Inside ambient (°F $\pm$ 1.4°) Heating air (°F $\pm$ 1.4°)

Test Procedures

The heat pump performed all necessary heating in the equipment storage and fabrication building during the winter of 1975-76. Tests on this unit were performed at steady state heat load, and ranged from 30 minutes to 90 minutes duration. Water flow rate and supply temperature were constant through each test. At all times, sufficient supply water was circulated to keep the evaporator above 40°F (4.4°C). The following parameters were also monitored and recorded for the duration of each test:

- temperatures at indicated points in the system (Fig. 7)
- amperage and voltage for the compressor motor
- amperage and voltage for the fan motor
- amperage and voltage for the entire machine.

This information made it possible to optimize the performance of the unit and analyze the cycle (ref. 10). For each test, an operating range was imposed by fixing the air temperature passing through the condenser. A test run was terminated when this temperature increased. High and low refrigerant pressures were monitored and recorded to yield directly the operating limits of the cycle.

Analysis of Performance

In addition to the economic evaluation which may be made once the CP has been calculated, a comparison of test runs demonstrates important characteristics of heat pumps. Table II presents information obtained from each test.

TABLE II. Summary of results of performance tests.

Test	CP	Supply temp (°C)	ΔT (°C)	High/low pressure (psig)	ΔP(psi)
1	3.76	13.39	7.0	250/60	190
2 (void)					
3	3.64	15.26	7.2	250/60	190
4	3.73	13.20	6.4	225/60	165
5	3.94	13.20	6.6	200/59	141
6	4.06	15.55	7.2	230/62	168
Avg CP	3.8				

Tests 1 and 4 operated with nearly the same coefficient of performance. Although test 1 had warmer supply water and therefore greater capacity for heat transfer in the evaporator, the pressure difference was greater than in test 4, and this disadvantage effectively canceled the effect of the warmer supply. Test 3 had the benefit of 59°F (15.3°C) supply water, as did test 6. But test 6 operated with a lower pressure difference than did test 3. Test 6 had the higher CP. Although the supply water temperature is the same for tests 4 and 5, the lower ΔP in test 5 yielded a higher CP.

The CP of test 6 was higher than that of test 5. This seems to contradict the previous analysis because the ΔP for test 6 is the greater of the two. Part of the explanation lies with the increased water supply temperature but the major contribution was the high suction pressure, which shifted the operating limits. Although the pressure difference is emphasized by the water pumping analogy, it is clear that the operating range is critical to optimum performance of a heat pump.

Economic Analysis

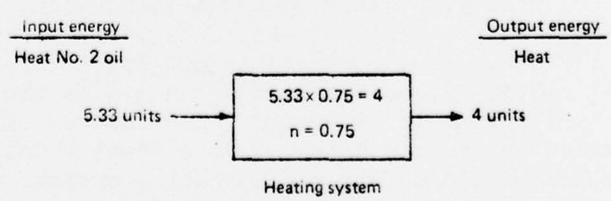
By definition of the coefficient of performance, the heat pump operations resulted in 3.5 to 4.0 times more heat output than the heat

available directly from the electrical input. This seems straightforward and very attractive. Electrical energy, however, is more costly than fuel energy. A careful analysis, therefore, is necessary for effective economic evaluation.

A well-maintained household furnace can operate with 75% steady state efficiency. This efficiency will be used in comparison with the heat pump operation. A gallon ($0.004\ m^3$) of number 2 fuel oil contains 140,000 Btu or 41.0 kWh (1.5×10^8 J). Delivery of this energy by a system 75% efficient yields 105,000 Btu (1.1×10^8 J), the energy equivalent of 30.8 kWh. Costs for this common fuel purchased in bulk are now about 40¢/gal (10.6¢/ℓ) or 1.3¢/kWh delivered at 75% efficiency. Electrical power cost was 3.7¢/kWh during the test period. Thus, for the same heating job, it is nearly three times as expensive to buy electricity for resistance heaters as it is to burn oil. How do heat pumps compare?

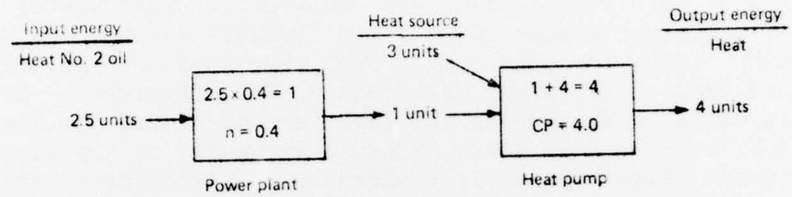
For an energy consumer to "break even," the heat pump he installs must operate with a CP high enough to at least equal the cost of operating a typical oil-fired unit. With a higher CP, of course, the consumer realizes a savings. The cost for operating the oil-fired system, established previously, is equivalent to 1.3¢/kWh. Buying electrical energy at 3.7¢/kWh means that the heat pump only "breaks even" with the oil system if it returns $3.7/1.3 = 2.8$ units of energy for each unit of energy purchased. Since the output/input ratio is the definition of the coefficient of performance, the minimum acceptable CP for this situation is 2.8. At an average CP of 3.8 for the machine used in the demonstration project, the cost of the heating was 0.97¢/kWh. This represents a 25% reduction in cost compared to heating with oil.

Being mindful of the second law of thermodynamics, we must always consider putting energy to the use which minimizes losses. The electricity which drove the heat pump to produce heat was generated, as usual, at a power station which burns fuel to produce heat to generate electricity. Each of these energy conversions has associated with it unrecoverable energy. It is therefore necessary to determine the minimum acceptable CP of a heat pump with respect to energy conservation, not only the economic break-even point discussed above. The 75% efficient oil-fired heating system capable of a specific task, say providing four units of heat, may be represented as follows:

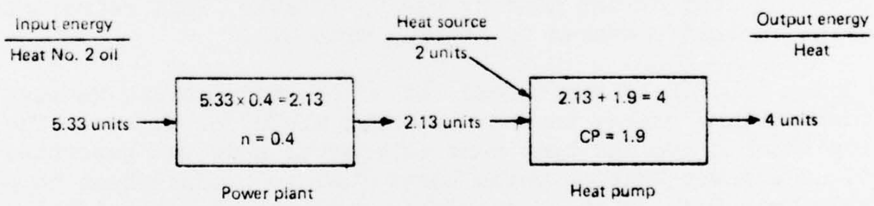


It "costs" 5.33 units of oil to produce 4 units of heat with this simple system.

Compare the above with the energy delivery and heating system for a heat pump. The typical thermal power plant is 40% or less efficient:



It costs 2.5 units of input oil to produce 4 units of heat. For the same heating, the second system requires less oil to be burned. The energy break-even performance level can be obtained by determining the CP for a system in which 5.33 units of oil input are necessary to produce 4 units of heating, as for the simple oil burner system:



The last diagram shows that a heat pump with CP = 1.9 uses as much fuel at the power source as would be used by a fired heating system at the delivery site. The U.S. Army Corps of Engineers, however, uses a "break-even" criterion which is based upon a source energy conversion into electricity, at less than 30% efficiency, applicable to typical fossil fueled power plants and transmission losses. For each kilowatt-hour (3415 Btu/h) delivered, 11,600 Btu (1.2×10^7 J) must be expended. This ratio, 11,600 Btu/3415 Btu = 3.4 is the practical minimum CP for a heat pump system to be energy conservative. This requires a higher performance level than a heat pump system, which is merely less expensive to operate than a fuel-fired heating system, for which the minimum CP was established previously to be 2.8.

CONCLUSIONS

Monitoring the performance of the heat pump revealed that the operating hours for a heat pump can be expected to far exceed the operating time required by the firing unit of a fueled heat plant, because the former is closely matched to the load, while fuel burning systems are usually oversized. It is important, therefore, to procure high quality components for the system because it will operate more hours per year than a furnace. Along this line, the CP is greatly affected by compressor and motor efficiencies, since this is where the costly input energy is used. The higher the efficiency of the motor-compressor, the more economical the operation of the heat pump.

Operating conditions for the heat pump system should be carefully identified. The equipment commercially available today operates with CP's in the range of 3.0 to 4.0 under the temperature conditions encountered in this demonstration project, with most systems under 3.5 (ref. 10). A heat pump system must be matched carefully to the conditions under which it will operate. Unlike a fuel-fired heating system, a heat pump operates <u>less</u> efficiently, therefore producing <u>less</u> heat, as the temperature of the heated space approaches the lower temperature limit of the operating range. This effect, demonstrated in tests 4 and 5, occurs because the suction pressure begins to fall off, reducing compressor efficiency. The absolute minimum heat output of this type of heat pump is the equivalent of the electrical input. This should not occur in a real system because it implies that no refrigeration is being done, and protection devices on all standard units would normally shut off electrical power to protect the refrigeration components.

Since the efficient performance range of heat pumps is so narrow, they are not well-suited to applications where room temperatures vary often and widely. The maintenance building in which the demonstration project was run received frequent blasts of frigid air when its large bay doors were opened. The heat pump required 4-6 hours after a normal day's traffic to stabilize the building temperature and cycle off at the thermostat. However, normal load through the winter was effectively handled by the unit, and space heating was provided for this building at 0.97¢/kWh (2.7×10^{-7} ¢/J) or $2.84/MBTU. A conservative estimate of costs for heating with oil is 1.3¢/kWh. Efficient operation of this system throughout the heating season resulted in a 25% reduction in heating costs, yielding significant savings. Based on current energy costs, when the CP of a heat pump is greater than 3.4, it is less costly to operate, both in terms of fuel and money. For this project, a CP of nearly 4.0 indicates a savings.

RECOMMENDATIONS

1. Analysis of published information should be made during system design to obtain expected performance information, particularly compressor efficiency.

2. This analysis should include not only an estimate of the heating load, but also whether the demand and space conditions are expected to be constant.

3. Future demonstration projects should show results for prolonged heating periods with various systems to provide comparison of maintenance costs and realistic "in service" operating characteristics, especially in comfort heating situations.

4. Installation and operation of conventional packaged units in heat pump applications should be considered economically favorable and energy conservative if their coefficient of performance exceeds 3.40.

APPENDIX A. SAMPLE CALCULATIONS

The following sample calculations demonstrate the use of the pressure-enthalpy diagram to accurately estimate the CP of a heat pump. Using published information on operating characteristics, the system designer may determine if a unit will produce heat efficiently and economically within the operating range to be imposed.

First, the results of Test 1, run on 23 March 1976, were used in this example to determine the real CP of the actual apparatus, and provide comparison with the estimated or reported CP.

Test 1

			Time
Elect. meter	Stop	13859.2	1350:00
	Start	-13855.1	-1310:00
		4.1 kWh	40 minutes elapsed time

1. Find total electrical power consumption of the entire machine for 1 hour.

   4.1 kWh for 40 minutes
   (60/40) (4.1) = 6.15 kW Total Power

2. Find fan motor power (rated at 1 hp or 0.7457 kW.
   Power factor = pf (due to 3ϕ source)   Measure 200 volts
   4.15 amps
   830 watts   Fan Power

   Worse Case pf = 1.0

3. Compressor electrical power consumption.   6150 W   Compressor Power
   -830 W
   5320 W

4. Find heat transfer of water

			Time
meter	Stop	522623.0	1400:00
	Start	-522210.0	-1315:15
		413.0 gal	44:45

$$\frac{413 \text{ gal}}{44.75 \text{ min}} = 9.23 \text{ gpm}$$

(9.23 gpm) (60 m/h) (8.345 lb/gal) = 4620 lb/h
1 lb H_2O requires 1 Btu/°F = 4620 Btu/h/°F available

5\. Water temp (evaporator)

	Supply	ΔT	Discharge
	13.6°C	6.9°C	6.7°C
	13.0	6.6	6.4
	12.9	6.9	6.0
	13.0	6.9	6.1
	13.5	7.1	6.4
	13.7	7.0	6.7
	14.0	7.3	6.7
(Avg. =	13.39)		

Sum ΔT = 48.7 Average ΔT = 6.96°C

6.96°C = 12.53°F

6\. From steps 4 and 5, find heat input to evaporator

(4620 lb/h) (1 Btu/h/°F) (12.53°F) =

57,890 Btu/h

or

(57,890 Btu/h) (2.931×10^{-4} kWh/Btu) = Heat Input of 16.97kW

7\. Find CP.

$$CP = \frac{\text{Total Heat Out}}{\text{Heat In}} = \frac{\text{Compressor Heat + Evaporator Heat}}{\text{Compressor Heat}}$$

From steps 3 and 6

$$CP = \frac{5.32\ \text{kW} + 16.97\ \text{kW}}{5.32\ \text{kW}} \approx 4.19$$

8\. Find CP (including fan motor)

$$CP\ \text{w/fan} = \frac{16.97\ \text{kW} + 6.15\ \text{kW}}{6.15\ \text{kW}} = 3.76$$

Steps 9-14 given below provide a check on the results of steps 1-8 and demonstrate a method of predicting the CP using manufacturers' data and the operating conditions for the system. Suction and discharge pressures may be determined from charts or tables for given temperature ranges.[12] For this test, suction pressure was held constant at 76 psia (5.2×10^{2} kPa) and discharge pressure was held at 260 psia (1.8×10^{3} kPa). An idealized thermodynamic cycle between these pressure limits is shown in Figure 3. Losses during each phase of the real cycle, particularly during nonisentropic compression, have an adverse effect on CP. Typically, these losses are not known, but may be measured, as in the previous steps, or estimated from detailed information available from manufacturers.

9. Enthalpy (h) values for the points of interest on Figure 3 are:

$$h_1 = 108 \frac{\text{Btu}}{\text{lb}}$$

$$h_2 = 122 \frac{\text{Btu}}{\text{lb}}$$

$$h_3 = 46 \frac{\text{Btu}}{\text{lb}}$$

$$h_4 = 46 \frac{\text{Btu}}{\text{lb}}$$

10. Determine compressor work and CP of this cycle.

Ideal compressor work = $h_2 - h_1$

(from step 9) 122-108 = 14 Btu/lb (3.3×10^4 J/kg)

$$\text{Ideal CP} = \frac{\text{condenser output}}{\text{compressor input}} = \frac{h_2 - h_3}{h_2 - h_1}$$

$$\text{(from step 9)}\ \frac{122-46}{122-108} = \frac{76\ \text{Btu/lb}}{14\ \text{Btu/lb}} = CP_{ideal} = 5.4$$

11. Estimate CP, using assumed or reported compressor efficiency (η) = 0.75.

$$\text{Estimated compressor work} = \frac{\text{ideal compressor work}}{\eta}$$

$$\frac{14}{0.75} = 18.7\ \text{Btu/lb}\ (4.3 \times 10^4\ \text{J/kg})$$

(from step 10 and 11)

$$CP_{estimate} = \frac{\text{condenser heat}}{\text{estimated compressor heat}} = \frac{76}{18.7}\ \frac{\text{Btu/lb}}{\text{Btu/lb}} = 4.06$$

12. Find real compressor heat.

Given: (from step 3) measured compressor power requirement = 5320 W

5320 W = 18,200 Btu/h.

(from step 10) ideal compressor does work at 14 Btu/lb (3.3×10^4 J/kg)

Assume: throttle at constant enthalpy

Points 1 and 3, (Fig. 3) apply to the real cycle

(from step 6) heat lost by water at evaporator = 17×10^3 W = 5.8×10^4 Btu/h

(from Fig. 3 and step 9) heat gained by R-22 at evaporator =

$h_1 - h_4 = 108$ Btu/lb $- 46$ Btu/lb $= 62$ Btu/lb (1.4×10^5 J/kg)

R-22 flow rate $= \dfrac{5.8 \times 10^4 \text{ Btu/h}}{62 \text{ Btu/lb}} = 935$ lb/h (0.12 kg/s)

(Real compressor heat $= \dfrac{18{,}200 \text{ Btu/h}}{935 \text{ lb/h}} = 19.5$ Btu/lb (4.5×10^4 J/kg)

13. Determine real compressor efficiency,

(from step 10) ideal compressor heat = 14 Btu/lb (3.3×10^4 J/kg)

(from step 12) real compressor heat = 19.5 Btu/lb (4.5×10^4 J/kg)

$$\eta_{real} = \frac{14 \text{ Btu/lb}}{19.5 \text{ Btu/lb}} = 0.72$$

14. Determine CP_{real} (from 10 and 12)

$$CP_{real} = \frac{\text{condenser heat}}{\text{real compressor heat}} = \frac{76 \text{ Btu/lb}}{19.5 \text{ Btu/lb}} = 3.90$$

Compare this with $CP_{estimate}$ (step 11) = 4.06

APPENDIX B. ERROR ANALYSIS.

To determine experimental uncertainty, a method demonstrated by Holman in Experimental Methods for Engineers[8] was utilized. Given the definition of coefficient of performance as

$$CP = \frac{\text{energy input} + \text{energy output}}{\text{energy input}}$$

note that the energy output for this case is refrigeration capacity. The numerator of the equation identifies the heat of rejection of the condenser, which heats the room, and the denominator is the electrical energy required to operate the system at some cost. The equation may be written

$$CP = 1 + \frac{\text{refrigeration effect}}{\text{electrical input}}$$

$$\text{or} \quad CP = f\,(W,\ gpm,\ \Delta T)$$

where CP = coefficient of performance, R

k = watts of electrical input to machine in 1 hour

G = gallons of water per minute through machine

T = temperature difference between supply and discharge water

1. To permit the above equation to be written in appropriate dimensional terms, we must insert

$$c = 1 \quad \frac{\text{Btu}}{\text{lb} \cdot {}^{o}\text{F}}\text{, the specific heat of water,}$$

$$\text{and} \quad c_1 = \left(8.345\ \frac{\text{lb}}{\text{gal}}\right)\left(17.59 \quad \frac{\text{watt}}{\text{Btu/min}}\right)$$

$$= 146.79 \quad \frac{\text{lb} \cdot \text{watt} \cdot \text{min}}{\text{gal} \cdot \text{Btu}}\text{, a constant.}$$

Then we can write the equation as

$$R = 1 + \frac{G\left(\frac{\text{gal}}{\text{min}}\right) \cdot T\ ({}^{o}\text{F}) \cdot c\left(\frac{\text{Btu}}{\text{lb}\ {}^{o}\text{F}}\right) \cdot c_1\ \left(\frac{\text{lb} \cdot \text{watt} \cdot \text{min}}{\text{gal} \cdot \text{Btu}}\right)}{k(\text{watt})}$$

$$= 1 + \frac{GTcc_1}{k}$$

2. We will need to have partial differentials of R with respect to the three variables:

$$\frac{\partial R}{\partial G} = \frac{Tcc_1}{k}$$

$$\frac{\partial R}{\partial T} = \frac{Gcc_1}{k}$$

$$\frac{\partial R}{\partial k} = GTcc_1 \cdot -k^{-2}$$

$$= -\frac{GTcc_1}{k^2} .$$

3. Now according to Holman, the uncertainty in R (U_R) is related to the uncertainty for each parameter (U_G, U_T, U_k) in the following way:

$$U_R = \left[\left(\frac{\partial R}{\partial G}\ U_G\right)^2 + \left(\frac{\partial R}{\partial T}\ U_T\right)^2 + \left(\frac{\partial R}{\partial k}\ U_k\right)^2\right]^{1/2} .$$

4. We want to evaluate Holman's equation using data from test 6:

$$R = 4.06 \pm U_R$$

$$G = 9.96 \pm 0.05\ \frac{gal}{min}$$

$$T = 13.01 \pm 1.4^{\circ}F$$

$$k = 6120 \pm 100\ watt$$

$$c = 1\ \frac{Btu}{lb \cdot {}^{\circ}F}$$

$$c_1 = 146.79\ \frac{lb \cdot watt \cdot min}{gal \cdot Btu}$$

Thus the terms in Holman's equation are:

$$\frac{\partial R}{\partial G} = \frac{Tcc_1}{k} = \frac{(13.01\ {}^{\circ}F)\ (1\ \frac{Btu}{lb \cdot {}^{\circ}F})\ (146.79\ \frac{lb \cdot watt \cdot min}{gal \cdot Btu})}{(6120\ watt)} = 0.312\ \frac{min}{gal}$$

$$\frac{\partial R}{\partial T} = \frac{Gcc_1}{k} = \frac{(9.96\ \frac{gal}{min})\ (1\ \frac{Btu}{lb \cdot {}^{o}F})\ (146.79\ \frac{lb \cdot watt \cdot min}{gal \cdot Btu})}{(6120\ watt)} = 0.239\ \frac{1}{{}^{o}F}$$

$$\frac{\partial R}{\partial k} = -\frac{GTcc_1}{k^2} = -\frac{(9.96\ \frac{gal}{min})\ (13.01^{o}F)\ (1\ \frac{Btu}{lb \cdot {}^{o}F})\ (146.79\ \frac{lb \cdot watt \cdot min}{gal \cdot Btu})}{(6120\ watt)^2}$$

$$= -\ 0.000508\ \frac{1}{watt}$$

$$U_G = 0.05\ \frac{gal}{min}$$

$$U_T = 1.4^{o}F$$

$$U_k = 100\ watt$$

5. Therefore, Holman's equation is evaluated as follows:

$$U_R = \left[(0.312\ \frac{min}{gal} \times 0.05\ \frac{gal}{min})^2 + (0.239\ \frac{1}{{}^{o}F} \times 1.4^{o}F)^2 + (-0.000508\ \frac{1}{watt} \times 100\ watt)^2 \right]^{1/2}$$

$$= \left[(0.0156)^2 + (0.335)^2 + (-0.0508)^2 \right]^{1/2}$$

$$= (0.0002 + 0.112 + 0.0025)^{1/2}$$

$$U_R = 0.339$$

6. Thus the uncertainty in R or CP is ± 0.34, so that for test 6,

$$3.72 \leq CP \leq 4.40.$$

It is interesting to note that the uncertainty associated with the water temperature measurements accounts for nearly the entire uncertainty in the coefficient of performance. If U_G and U_k were both zero, U_R would still be as large as 0.335, compared to the value computed above of 0.339. It is also apparent that the uncertainty in the water flow measurements has a negligible contribution to U_R.

Results of this analysis show that the range of possible values of CP, from 3.72 to 4.40, represents a potential error of $\pm$ 8%.

LITERATURE CITED

1. Aamot, H. W.C. (1974) Management of power plant waste heat in cold regions. CRREL Technical Report 257, AD A993217.

2. Aamot, H.W.C. (1976) Central heat pump heating system for Ice Engineering Facility, CRREL Technical Note (unpublished).

3. Ambrose, E.R. (1966) Heat pumps and electric heating. New York: John Wiley & Sons, Inc.

4. ASHRAE Guide and Data Book (1966) Applications. American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc., New York, Chapter 6.

5. ASHRAE (1967) Handbook of fundamentals. American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc., New York, p. 299-301.

6. Crosby, R.L., H.W.C. Aamot, E.A. Wright (1975) Thermal energy and the environment. CRREL paper presented at Energy and Environment Conference, Dallas.

7. Freund, P. (1975) Heat pumps. Building Research Establishment, Garston, England, p. 6-7.

8. Holman, J.P. (1971) Experimental methods for engineers. 2nd ed., New York: McGraw-Hill. p. 37-44.

9. King, G. R. (1965) Basic air conditioning. Chicago: Nickerson & Collins Co., Chap. 14.

10. Lovell, T.W. (1976) Increasing heat pump performance. B.E. Project Report, Thayer School of Engineering, Dartmouth College. (unpublished).

11. National Building Code (1967) Air conditioning appliances and heat pumps. Engineering and Safety Department, Section 12.

12. Richter, H. J. (1976) Cost effective space heating with low temperature heat. Thayer School of Engineering, Dartmouth College.

13. Strock, C. and R. L. Koral (1965) Handbook of air conditioning, heating and ventilating. New York: The Industrial Press, Sect. 1-32, 1-137.